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**COMPARATIVE PERFORMANCES OF TWO WARM-AIR
PERIMETER SYSTEMS AND THREE CONVECTION SYSTEMS**

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UNIVERSITY OF ILLINOIS BULLETIN

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**COMPARATIVE PERFORMANCES OF TWO WARM-AIR PERIMETER
SYSTEMS AND THREE CONVECTION SYSTEMS**

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ABSTRACT

This bulletin is the first report of results obtained in Warm-Air Heating Research Residence No. 3, which is a low-cost basementless house with a concrete slab floor laid on the ground. The Residence was built by the National Warm Air Heating and Air Conditioning Association in order to provide a facility for comparative studies of the performances of several warm-air heating systems which appeared to be suitable for heating basementless houses having concrete slab floors.

The results are given for an investigation of five types of warm-air heating systems, including:

1. a two-loop perimeter system
2. a radial-feeder perimeter system
3. a conventional forced-air system with low-wall registers
4. a conventional forced-air system with high-wall registers, and
5. a gravity system.

The two perimeter systems consisted of 8-in.-diam sheet metal ducts embedded in the concrete slab at the periphery of the house so as to enclose practically the entire floor area. Warm air was supplied to the perimeter duct through 8-in.-diam feeder ducts which extended from a subfloor plenum below the furnace to the perimeter duct. The air supplied to the perimeter duct entered the rooms through floor registers which were located below the windows.

The results obtained with the three convection systems were not entirely satisfactory. The room-air temperature differentials from floor to ceiling were large for the high-wall and gravity systems; room-air temperatures at the 30-in. level were not uniform for the low-wall system; and floor-surface temperatures were low in the areas near the outside walls for all three systems.

The performance characteristics for the two-loop perimeter system were generally superior to those for the convection systems. Excellent control of room-air temperatures was provided by the use of a conventional room thermostat. However, some difficulty was experienced in balancing the system and in maintaining satisfactory floor-surface temperatures in the bedrooms. Analysis of the performance data indicated that the use of a continuous perimeter duct in conjunction with properly spaced feeder ducts would improve results. Hence an investigation of a

ABSTRACT (Concluded)

radial-feeder perimeter system was undertaken to determine the optimum number and location of feeder ducts.

Four arrangements of feeder ducts were studied in the investigation of the radial-feeder perimeter system—two arrangements with three feeders each, a four-feeder arrangement, and a five-feeder arrangement. The results obtained with the two arrangements in which feeder ducts were extended into the corners of the Residence were much better than those obtained with the two-loop system; the room-air temperature differentials were markedly reduced, the cold areas on the floor surface were virtually eliminated, and no difficulty was experienced in balancing the system.

On the basis of the results obtained in this investigation, the radial-feeder perimeter system with feeder ducts extending into exposed corners was the most satisfactory system for heating the Residence.

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I. INTRODUCTION

1. Preliminary Statement

This bulletin is the first report of results obtained in Warm-Air Heating Research Residence No. 3 under the terms of a cooperative agreement between the National Warm Air Heating and Air Conditioning Association and the University of Illinois. The Residence, a low-cost basementless house with a concrete slab floor laid on the ground, was constructed in 1949 and was equipped specifically for research in warm air heating. Construction of this Residence was an outgrowth of the urgent need for information on the performances of warm-air heating systems which appeared to be suitable for heating basementless houses that are built with concrete slab floors.

During the period of this investigation the Association was represented by a Research Advisory Committee of sixteen men:

F. L. Meyer, Chairman, The Meyer Furnace Co., Peoria, Ill.

R. K. Becker, Ohio Valley Hardware and Roofing Co., Evansville, Indiana

J. B. Burrowes, Lau Blower Co., Dayton, Ohio

K. T. Davis, Bryant Heater Division of Affiliated Gas Equipment, Inc., Cleveland, Ohio

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F. W. Taylor, Canadian Chapter, National Warm Air Heating and Air Conditioning Association, Toronto, Ontario, Canada

H. Weyenberg, Holland Furnace Co., Holland, Michigan

2. Acknowledgments

Acknowledgment is made to the manufacturers who cooperated by furnishing equipment used in the investigation. The assistance of J. E. Sjoldal and G. W. Sadler, former graduate students, and J. W. Brill, Research Assistant, is gratefully acknowledged.

3. Scope of Investigation

The tremendous increase in the construction of basementless houses during recent years has introduced new problems in connection with the heating of these homes. The construction of homes provided with a concrete slab floor laid on the ground has introduced such difficulties as cold floor surfaces and condensation on these surfaces.^{(1)*} The problems are accentuated when a slab floor is used with a low-cost house which is just sufficient to comply with minimum construction standards. The use of conventional warm-air heating systems has not solved the problem of cold floors. Hence, many new approaches have been suggested⁽²⁾ and some of these have given satisfactory performance in field installations. One such approach is a panel-convection system designated as the warm-air perimeter heating system. To obtain performance data as a guide for establishing a design procedure, the investigation in Residence No. 3 was devoted largely to various modifications of this new system. For purposes of comparison with the more conventional warm-air heating systems, three convection systems were also studied.

This bulletin reports the performance obtained with the following heating systems:

1. a two-loop warm-air perimeter system
2. a radial-feeder warm-air perimeter system
3. a conventional forced-air system with low-wall registers
4. a conventional forced-air system with high-wall registers, and
5. a gravity system.

During the period of this investigation a simultaneous study was being conducted in the Floor Slab Laboratory at the University of Illinois on the thermal characteristics of a floor slab heated by means of an embedded perimeter duct. The results from the Laboratory were correlated with those from the perimeter systems which were investigated in Residence No. 3.

* Parenthesized superscript numerals refer to the correspondingly numbered entries in the References.

4. Glossary

Air-flow rate — The rate of circulation of air in cu ft per min (cfm).

Unless otherwise stated, all cfm values are for standard air density of 0.075 lb per cu ft.

Balance of room-air temperatures — Uniformity in room-air temperatures between different rooms served by a single room thermostat, as measured at the 30-in. level.

Blower — A centrifugal fan. The warm-air heating industry uses the term to distinguish centrifugal fans from propeller fans.

Bonnet efficiency — The ratio of the bonnet capacity to the heat liberated in the furnace by the burner, also expressed as a percentage. For gas-fired forced-air furnaces approved by the American Gas Association the rated bonnet efficiency is 80 percent.

Breathing level — Level in room 60 in. above floor.

Ceiling level — Level 3 in. below ceiling.

Continuous blower operation — A method of blower operation in which continuous operation is approached in average winter weather but intermittent operation is obtained in mild weather.

Design heat loss — The calculated heat loss for a given space based on outdoor design conditions for the locality. In the text the outdoor design conditions are assumed as -10°F and 15 mph wind velocity.

Floor level — Level in room 3 in. above floor.

Fuel consumption — The consumption of fuel per 24 hr. For gas-fired equipment the units are in terms of cu ft of gas per 24 hr.

Fuel-input rate — The rate of heat liberation in the furnace by the burner expressed in Btu per hr.

Furnace bonnet — A central plenum, or collecting chamber, located usually above the furnace, in which the heated air is mixed before distribution to the duct system. For perimeter systems using a down-flow furnace the subfloor plenum is equivalent to a bonnet.

Indoor-outdoor temperature difference — The difference in temperature between indoor air and outdoor air. Large temperature differences denote cold weather; small temperature differences indicate mild weather.

Isothermal lines, floor surface — Lines of equal floor-surface temperature.

Living zone — The space in a room between the floor level and breathing level.

Panel heating effect — A heat transfer effect similar to that obtained from a panel heating system, in which warmed surfaces transmit heat

by radiation to cooler surfaces and by convection to cooler air next to the panel surfaces.

Register delivery — The heat available at the registers, in Btu per hr. This is based on the air-flow rate through the registers and the difference between register-air temperature and the air temperature at the return-air intake.

Sitting level — Level in room 30 in. above floor.

Temperature differential, room-air — The difference in air temperature in a room at two elevations. Usually the sitting level, 30 in. from floor, is considered as the reference level. See *Temperature gradient*.

Temperature gradient, room-air — A representation of the air temperatures existing at several levels in a room at one station. See *Temperature differential*.

Thermostat differential setting — An adjustable setting in the room thermostat which governs the degree of fluctuations in room-air temperature at the thermostat.

II. EQUIPMENT AND GENERAL PROCEDURE

5. Research Residence No. 3

Residence No. 3 (Fig. 1) was a single-story, low-cost home with a concrete slab floor, was of standard frame construction, and was provided with a vented attic and a relatively large amount of glass area. The walls were uninsulated but the ceiling was insulated with mineral wool batts $3\frac{5}{8}$ in. thick. Except for one picture window, well-fitted double-hung wood-sash windows were used. The windows were not weatherstripped nor were they equipped with storm sash. The outside doors, however, were provided with storm doors.



Fig. 1. Warm-Air Heating Research Residence No. 3

The slab floor construction (Fig. 2) consisted of a 4-in. gravel fill placed on the original grade, a heavy duplex paper vapor barrier (damp-proofing membrane) tarred at the joints, and 4 in. of concrete. The insulation at the edge of the slab consisted of a $2\frac{5}{32}$ -in. asphalt-coated fiber insulating board placed against the foundation wall and extending downward 12 in. from the top of the slab.

A floor plan of the Residence is shown in Fig. 3. The inside dimensions were 24 ft x 32 ft and the corresponding floor area was 768 sq ft. The design heat loss of the Residence for an outdoor temperature of -10°F and an indoor temperature of 70°F was about 51,600 Btu per hr for

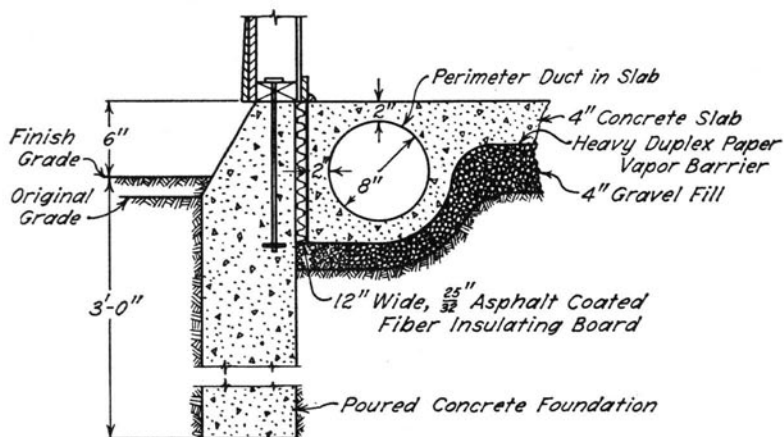


Fig. 2. Details of Floor Slab Construction Showing Perimeter Duct

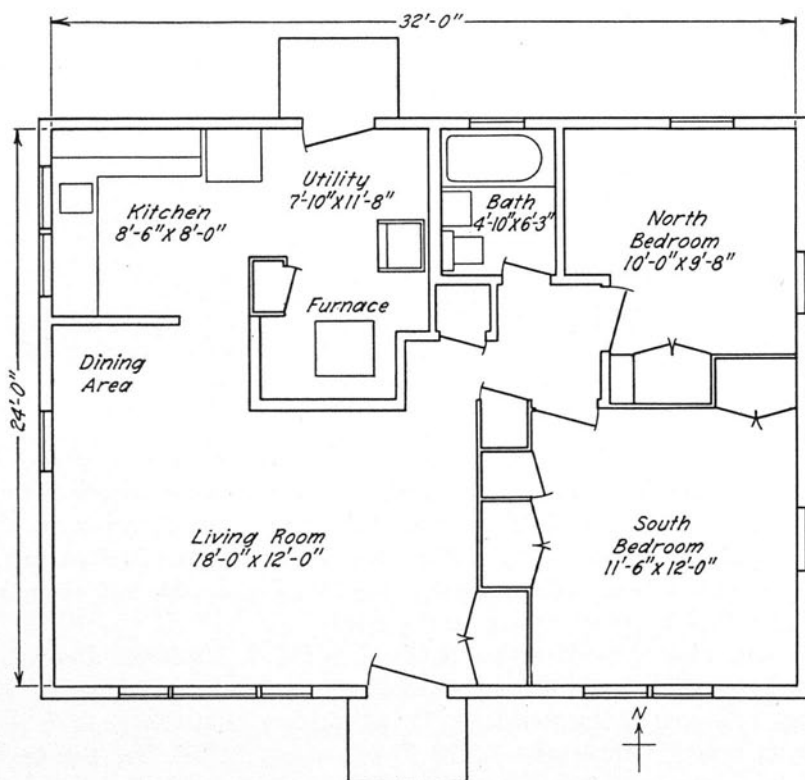


Fig. 3. Floor Plan of Research Residence No. 3

operation with the perimeter system and about 48,800 Btu per hr for operation with the convection systems. A tabulation of heat transmission coefficients and room dimensions is shown in Table 1.

6. Heating Systems Investigated

The four heating systems investigated during the 1949-50 heating season included a two-loop perimeter system, a forced-air system with low-wall registers, a forced-air system with high-wall registers, and a gravity system. The heating plant used was of the hi-boy type and could be operated as either a gravity or a forced-air unit. The input rating of the furnace was 100,000 Btu per hr. The conversion from forced-air operation to gravity operation required changes in the furnace unit, warm-air registers, and return-air intakes. Those sections of the duct system which were used for both gravity and forced-air operation were sized for gravity operation and were larger than was necessary for forced-air operation alone.

During the 1950-51 heating season, four different arrangements of a radial-feeder perimeter system were studied. The furnace was of the down-flow type with the blower located below the heat exchanger. Its input rating was 65,000 Btu per hr.

7. Instrumentation

Approximately 250 copper-constantan thermocouples were used to determine temperatures. All thermocouples were connected to individual switches which in turn were connected to an indicating potentiometer. By means of two recording potentiometers, continuous printed records of the temperatures at any 22 thermocouple stations could be obtained.

The bonnet-air temperature was measured by means of a grid, consisting of six thermocouples connected in parallel, so located that it was not affected by direct radiation from the heat exchangers in the furnace. The flue gas temperature was measured by means of a recording thermometer. The relative humidity was recorded by an instrument located in the living room. Periodic readings were made with an Orsat analyzer of the CO₂ content in the flue gases before their entrance into the draft hood. The electrical inputs to the burner and to the blower motor were measured by watt-hour meters reading directly to 10 watt-hours. Self-starting electric clocks were connected across the burner and blower motor circuits, to obtain the total time of operation of each. The fuel input to the furnace was determined by means of a gas meter.

8. General Procedure

The thermostat, located at the 30-in. level in the living room, was set at 72 F. The differential setting of the thermostat was adjusted to the

Table 1
Data on Research Residence No. 3

Room	Room Dimensions	Floor Area, sq ft	Volume, cu ft	Calculated Heat Loss, Btu per hr			
				Above- Ground ^b	Below-Ground ^c	Perimeter in Use	Perimeter Not in Use
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
Living Room	18 ft 2 in. x 12 ft 0 in.						
Closet	8 ft 6 in. x 4 ft 0 in.	251	2 008	14 875	2 790	1 940	17 665
So. Bedroom	4 ft 0 in. x 2 ft 4 in.	9	72	(^d)	(^d)	(^d)	(^d)
Closet	12 ft 0 in. x 11 ft 6 in.	138	1 104	8 511	2 070	1 440	10 581
Closet	6 ft 0 in. x 2 ft 4 in.	14	112	(^d)	(^d)	(^d)	(^d)
No. Bedroom	4 ft 0 in. x 2 ft 4 in.	9	72	(^d)	(^d)	(^d)	(^d)
Closet	9 ft 8 in. x 8 ft 0 in.	92	736	6 468	1 863	1 300	8 331
Bath	6 ft 3 in. x 2 ft 3 in.	9	72	(^d)	(^d)	(^d)	(^d)
Utility	4 ft 0 in. x 2 ft 4 in.	34	272	2 729	450	314	3 179
Kitchen	12 ft 3 in. x 5 ft 5 in.	96	768	4 022	616	328	4 638
Hall	8 ft 4 in. x 8 ft 0 in.	67	536	5 864	1 360	945	7 224
Closet	7 ft 8 in. x 5 ft 8 in.	44	352	(^d)	(^d)	(^d)	(^d)
Closet	2 ft 4 in. x 2 ft 0 in.	5	40	(^d)	(^d)	(^d)	(^d)
Total		768	6 144	42 469	9 149	6 367	51 618
							48 836

^a The exterior wall consisted of a double course of cedar shingles, building paper, shiplap sheathing on 2-in. by 4-in. studs, 1/2-in. gypsum board with aluminum foil backing and a simulated plaster finish.

^b The heat losses above ground were calculated by methods given in the *Manual 303* of NWAHACA, 1949 ASHVE *Guide*, (^d) the crackage method being used to calculate the heat loss due to infiltration at the windows and doors.

^c The heat loss below ground was calculated on the basis of information contained in the University of Illinois Small Homes Council Report on "Temperature and Heat Loss Characteristics of Concrete Floors Laid on the Ground."^(e)

^d Heat loss for these rooms included with larger adjoining rooms.

^e Heat loss for these rooms included with north bedroom.

minimum point to provide frequent burner operations. All doors between rooms were kept open.

The fuel used was natural gas with a calorific value of 1000 Btu per cu ft. The desired fuel-input rate of 65,000 Btu per hr was obtained by adjusting the gas-flow rate at the meter. This input rate was determined by dividing the design heat loss (51,600) by the assumed bonnet efficiency (0.80) of the furnace. The design heat loss of the Residence when the perimeter system was not in operation was about 48,800 Btu per hr, indicating that an input rate of 61,000 Btu per hr would have been sufficient. However, in order to make a direct comparison of all performance factors, the input rate of 65,000 Btu per hr was used throughout the investigation.

During the 1949-50 heating season the air-flow rate through the furnace unit was determined indirectly⁽⁶⁾ by first subtracting the calculated values of flue and jacket losses from the input rate and then dividing this value by the product of the air temperature rise through the furnace and the specific heat of air. The resulting air-flow rate in lb per hr was converted to cfm. During the 1950-51 season the same procedure was followed; in addition, direct flow measurements were made by means of a calibrated vane anemometer⁽⁷⁾ located in the return-air duct.

Complete daily records were made of the operating times, the number of cycles, and the electrical inputs to both the blower motor and the burner. Daily observations were made of the fuel consumption of the furnace. Either periodic or continuous records of all significant temperatures were also made.

Comfort depends on a large number of factors such as air temperatures, surface temperatures, relative humidity, and air movement as well as a number of more subjective items such as odor, noise, and dust content. Since it was not possible to evaluate all these items, emphasis was placed on two predominant factors—room-air temperatures and floor-surface temperatures. The study of air temperatures was devoted primarily to the conditions obtained in the living zone between the floor level and the breathing level, since temperatures above the breathing level affect comfort but little.

III. PERFORMANCE OF TWO-LOOP PERIMETER SYSTEM

9. Description and Procedure

In this system, 8-in.-diam sheet metal ducts were embedded 2 in. below the top of the concrete slab, and were installed in the form of two loops which enclosed a major portion of the floor area (Fig. 4). The warm air was delivered downward from the bonnet of the hi-boy furnace through an insulated duct into a subfloor plenum below the furnace. From the plenum the air was forced outward to the perimeter ducts and then through these ducts to the registers. After entering the rooms at the registers the air moved across the rooms to the return-air intakes near the center of the house and then into the furnace to be reheated.

Experimental conditions for the various series of studies are listed in Table 2. Three main studies, designated as Series P-3, P-5, and P-6, were conducted. Except for the locations of the return-air intakes, the experimental conditions were the same for all three series. In all cases the system was operated in accordance with the continuous air circulation principle⁽⁸⁾ — that is, with relatively low cut-in and cut-out settings of the fan switch and with relatively low rates of air flow. Series P-1 and P-2 were preliminary studies in which the duct between the bonnet and the subfloor plenum was not insulated. Since the vagrant heat gain from the ducts in the utility room was found to be large, for the later series the ducts were insulated.

The study designated as Series P-4 was conducted with various arrangements of registers on the east loop in order to determine the effects of altering the air-flow rates in the different sections of the loop. For this purpose the alternate registers in the bedrooms were used instead of the north and south registers. In another phase of this study the bathroom register was closed.

10. Uniformity of Room-Air Temperatures at Sitting Level

The cyclical variation of room-air temperatures was small, amounting to about 0.5 F in the living room. Also, the response of the system to sudden changes in outdoor temperature was considered to be satisfactory, since no evidence of lag or overrun of room-air temperatures was obtained during periods of rapidly changing outdoor temperatures. Hence the conventional room thermostat was considered to be satisfactory for use with the perimeter system.

Preliminary observations indicated that some adjustment of the air flow rates to the various rooms was required in order to obtain uniform air temperatures. Since conventional dampers could not conveniently be installed in the duct system, the system was balanced by means of shutter dampers in the floor registers. Initial observations indicated that the temperatures in the bedrooms were about 3 F lower than those in

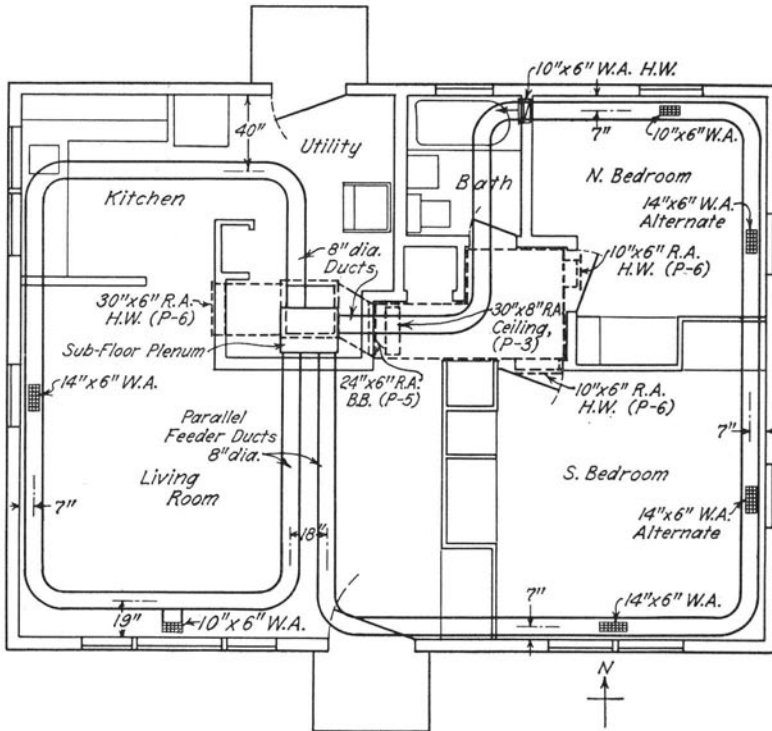


Fig. 4. Plan of Two-Loop Perimeter System

the living room. After adjusting the dampers in the living room registers until they were practically closed, the temperature of the south bedroom was about 1.0 F lower and that of the north bedroom was 1.5–2.0 F lower than that of the living room on a day during which the outdoor temperature was about 35 F and a moderate wind velocity prevailed. For an outdoor temperature of about 10 F the average difference between rooms with the low-wall return-air intake (Series P-5) was about 1 F, whereas with the ceiling return-air intake (Series P-3) and the high-wall return-air intakes (Series P-6) the differences were consistently 2–3 F.

Table 2
Experimental Conditions
 1949-1950 Heating Season

Series	Temp. Rise, F	Air- Flow Rate, cfm	Location of Registers				Locations of Return-Air Intakes				Period of Observation
			Liv. Rm.	S.B. Rm.	N.B. Rm.	Bath	Hall	Din. Area	S.B. Rm.	N.B. Rm.	
P-1* (Perimeter)†	100	480	Fl(W) Fl(S)	Fl(S)	Fl(N)	HW(NE)	Ceiling (W)				Nov. 16, 1949- Dec. 2, 1949
P-2* (Perimeter)†	100	480	Fl(W) Fl(S)	Fl(E)	Fl(E)	HW(NE)	Ceiling (W)				Dec. 3-Dec. 18, 1949
P-3 (Perimeter)†	100	480	Fl(W) Fl(S)	Fl(S)	Fl(N)	HW(NE)	Ceiling (W)				Dec. 19, 1949- Jan. 8, 1950
P-4 (Perimeter)†	100	480	Fl(W) Fl(S)	See text	See text	See text		HW(E)	HW(NW)	HW(SW)	Feb. 7-Feb. 11, 1950
P-5 (Perimeter)†	100	480	Fl(W) Fl(S)	Fl(S)	Fl(N)	HW(NE)	BB(W)				Jan. 9-Jan. 26, 1950
P-6 (Perimeter)†	100	480	Fl(W) Fl(S)	Fl(S)	Fl(N)	HW(NE)		HW(E)	HW(NW)	HW(SW)	Jan. 27-Feb. 6, 1950
L-1 (Low Wall)	100	480	LW	LW	LW(S)	HW(SW)		HW(E)	HW(NW)	HW(SW)	Feb. 17-Feb. 24, 1950 Apr. 7-Apr. 14, 1950
H-1 (High Wall)	70	680	HW	HW	HW(SW)	HW(SE)	BB(W)	BB(E)			Mar. 1-Mar. 13, 1950 Apr. 4-Apr. 6, 1950
G-1 (Gravity)	vari- able	vari- able	HW	HW	HW(SW)	HW(SE)	BB(W)	BB(E)			Mar. 14-Apr. 3, 1950

1950-1951 Heating Season

Series	Temp. Rise, F	Air- Flow Rate, cfm	Location of Registers				Locations of Return-Air Intakes	Period of Observation
			Liv. Rm.	S.B. Rm.	N.B. Rm.	Bath		
P-11 (Perimeter)†	96	500	Fl(W) Fl(S) Fl(S)	Fl(S) Fl(E)	Fl(N) Fl(E)	Base of Tub	Fl(NE)	Nov. 16-Dec. 11, 1950 Feb. 13-Feb. 25, 1951
P-12 (Perimeter)†	96	500	Fl(W) Fl(S) Fl(S)	Fl(S) Fl(E)	Fl(N) Fl(E)	Base of Tub	Fl(NE)	Dec. 15, 1950- Jan. 8, 1951
P-13 (Perimeter)†	96	500	Fl(W) Fl(S) Fl(S)	Fl(S) Fl(E)	Fl(N) Fl(E)	Base of Tub	Fl(NE)	Jan. 12, 1951- Feb. 9, 1951
P-14 (Perimeter)†	96	500	Fl(W) Fl(S) Fl(S)	Fl(S) Fl(E)	Fl(N) Fl(E)	Base of Tub	Fl(NE)	Feb. 26, 1951- Apr. 1, 1951

* Preliminary series—Duct from bonnet to

subfloor plenum not insulated

† Two-loop perimeter system

‡ Radial-feeder perimeter system

Fl = Floor register

HW = High-wall register

BB = Baseboard register

LW = Low-wall register

Operating conditions common to all series:

(a) Fuel input, Btu per hr = 65 000

(b) Thermostat setting = 72 F

(c) Fan switch settings for Series P-1 through H-1

Cut-in = 100 F

Cut-out = 80 F

(d) Limit switch setting for Series P-1 through H-1

Cut-out = 200 F

Cut-in = 185 F

(e) Limit switch setting for Series G-1

Cut-out = 225 F

Cut-in = 200 F

(f) House unoccupied

(g) No filters in unit

The fact that the bedroom-air temperatures were always lower than those for the living room indicated that the limit of adjustment had been reached. This difficulty in balancing the system was attributed to the panel heating effect from the living room floor, together with the low register temperatures in the bedrooms, as discussed in Section 12.

Observations of the temperatures at seven thermocouple stations in the living room at the sitting level indicated that the east half of the room was slightly warmer than the west half. The greatest difference, which was not more than about 2 F, occurred between the east area above the parallel ducts and the cooler northwest corner of the room. This satisfactory uniformity of air temperatures was maintained with all three series of studies over the entire range of weather conditions, and compared favorably with the results⁽⁷⁾ obtained in Research Residence No. 2, in which the walls were fully insulated and storm sash were used.

11. Room-Air Temperature Differentials

For any given heating system the room-air temperature differential from floor to ceiling depends largely on the outdoor weather conditions and on the accompanying operation of the system. The differential is also affected by extraneous heat gains from solar effects, cooking, bathing, etc., none of which originate with the heating system. In the case of Research Residence No. 1, the room-air temperatures observed at 7 a.m., 11 a.m., 4 p.m., and 10 p.m. were averaged for the day and the average values were plotted against the indoor-outdoor temperature difference for the same day. This method of showing average room-air temperatures was not satisfactory for Residence No. 3 because, as in the case of Residence No. 2, the solar heat effects⁽⁷⁾ were so pronounced that any differences attributable to the method of plant operation were largely obscured. Hence, as in the report of results obtained in Residence No. 2, the practice of showing only the 7 a.m. temperature gradients was adopted even though temperature observations made at 7 a.m. did not indicate the performance of the system in its more favorable aspects.

Figures 5a, 5b, and 5c show room-air temperature gradients for outdoor temperatures in the range of 10–15 F. The temperature gradients for a given area in a room were greatly affected by the location of the heat source and the manner in which the heat was introduced into the room. For example, Fig. 5a shows the gradients for the two bedrooms and the west half of the living room. In these areas, in which the heat sources were the floor registers and the warm floor above the perimeter

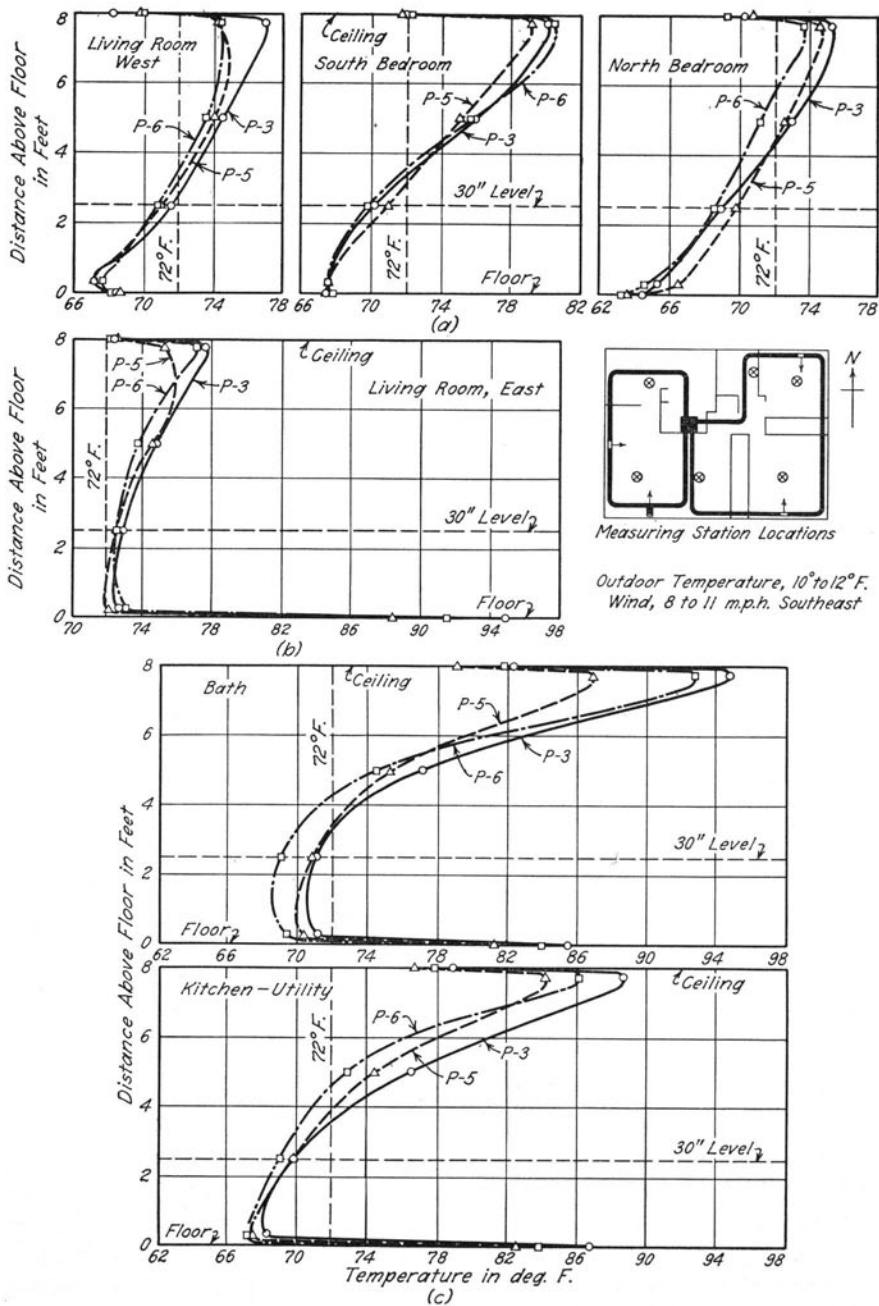


Fig. 5. Room-Air Gradients for Two-Loop Perimeter System

ducts, the temperature gradients observed at the measuring stations were governed largely by convection heating effects and to only a minor extent by panel heating effects. For the three series the average room-air temperature differentials were from 3—4.5 F between the floor level and the sitting level, and from 6—8.5 F between the floor level and the breathing level. The differentials for the three series were approximately the same, indicating that the location of the return-air intake had no appreciable effect on the differentials.

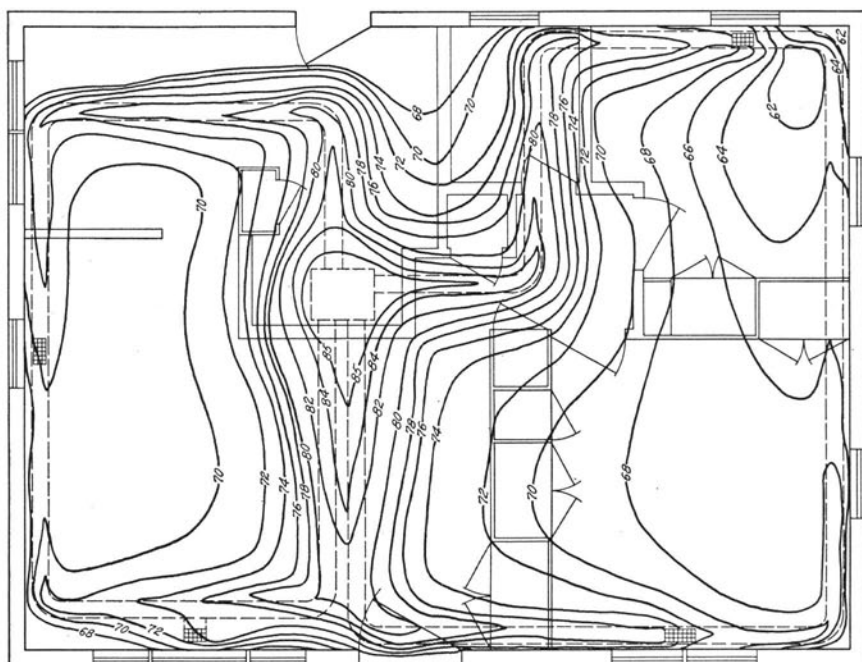
Figure 5b shows representative temperature gradients for the east half of the living room where the heat sources were the floor registers and the warm floor above the parallel ducts. The measuring station was located directly above the parallel ducts. Temperature differentials in this area were smaller than in any other area and reflected the panel heating effect created by the two ducts. For the three series the average differentials from the floor level to the sitting level were from -0.5 F to +0.5 F, and from 0.5 to 2.5 F between floor level and breathing level.

Temperature gradients for the bathroom and the kitchen-utility room, both of which had unusual heat sources, are shown in Fig. 5c. For example, the heat sources in the bathroom were the warm floor above the embedded duct and a high side-wall register. In the kitchen the heat sources consisted of the heated floor and the heated surfaces of the furnace casing, furnace bonnet, and flue. The resulting temperature differentials between the floor level and the sitting level were negligibly small, but large differentials existed between the sitting level and the ceiling level as a result of the stratification of warm air in the upper part of the room. Although the temperature differentials experienced with the two-loop system were not as small as had been expected, the analysis indicated that improvements could be made by proper spacing of the feeder ducts. The performance obtained with an improved perimeter system is discussed in Chapter V.

12. Floor-Surface Temperatures

The floor surface isotherms shown in Fig. 6 are representative of conditions obtained at the end of a 3-day period during which the outdoor temperature was about 40 F, cloudy weather prevailed, and the wind velocity was approximately 5 mph from the southwest. The floor-surface temperatures did vary somewhat with changes in outdoor temperature. For instance, as the outdoor temperature decreased, surface temperatures above the ducts increased as a result of higher duct-air temperatures. However, temperatures in areas away from the ducts remained essentially constant.

A concentrated source of panel heating effect existed in that portion of the living room floor above the parallel ducts, as well as in the floors of the utility room and hall, and undoubtedly contributed to the unbalance of the system. The conclusion was reached that the ducts leading to the perimeter should be spaced far apart and under separate rooms if possible.



Weather: No Sun; Outdoor Temp. 41.5°F ; Wind 5 mph (SW)

Fig. 6. Floor-Surface Isotherms with Two-Loop Perimeter System

The floor-surface temperatures near the center of the bedrooms were low, even though the area was bounded by the perimeter duct. This low-temperature area may be explained by the fact that the air in the perimeter duct was not maintained at a sufficiently high temperature. Most of the air flowing around the south side of the loop entered the south bedroom register, leaving only a small quantity to flow around the 30-ft length of the east loop to the north bedroom register. Hence a considerable temperature drop was obtained in this 30-ft length, resulting not only in a low register-air temperature but also in low floor-surface temperatures. The net result of the excessive temperature

drop was a deficiency of heat input to the north bedroom. Attempts to correlate the rate of temperature drop with distance along the duct were not successful. Since the many variables which affected temperature drop could not be separately evaluated in the Research Residence, a study was conducted at the Floor Slab Laboratory, as reported in separate papers.^(9,10,11)

The evidence shown in Fig. 6 indicated that a more favorable design would consist of a continuous perimeter duct with (a) relatively short distances between the feeder ducts and the registers, (b) the feeder ducts so arranged that all sections of the perimeter duct would serve as effective air passages, and (c) relatively high air-flow rates maintained in all sections of the perimeter duct.

13. Performance of Burner and Furnace

Although the up-flow furnace required a vertical duct connection between the furnace bonnet and the subfloor plenum, no difficulty was met in adapting it for use with the perimeter system. It was necessary, however, to insulate the vertical duct connection in order to minimize the vagrant heat gain into the kitchen-utility room.

The operating characteristics of the burner and furnace—including such items as hours of operation, electrical input to motors, number of operations per day, and fuel input—were essentially the same for the three series of studies. Such operating characteristics of the burner and furnace are presented in Chapter IV and compared with those of the three convection systems.

With the cut-in point of the fan switch set at 100 F and the cut-out point at 80 F, the blower operated practically continuously when the outdoor temperature was 50 F and lower. In order to conform with the continuous air circulation principle⁽⁸⁾ the fan speed was adjusted to provide a temperature rise through the furnace unit of 100 F. In Series P-5, in which the return air was drawn into the furnace at the floor level at about 70 F, the operating bonnet-air temperature attained a value of 170 F after prolonged burner operation. However, in Series P-3 and P-6, in which the return-air intakes were located at the ceiling level, the operating bonnet-air temperature attained a value of approximately 180 F after prolonged operation of the burner. From an analysis of the results it would appear that a lower bonnet-air temperature and a correspondingly larger air-flow rate would result in lesser panel effect from the feeder ducts close to the furnace, a lower rate

of temperature drop along the duct, and smaller room-air temperature differentials. However, the use of a temperature rise through the furnace unit considerably smaller than 100 F would result in furnace specifications which would be at variance with the existing rating requirements. The studies involving low temperature rises, therefore, were not made, particularly since the analysis also indicated that an alternative study involving a relocation of the feeder ducts seemed more promising. Such an alternative study of a radial-feeder perimeter system is reported in Chapter V.

IV. PERFORMANCE OF LOW-WALL, HIGH-WALL, AND GRAVITY SYSTEMS

14. Description and Procedure

In the last half of the heating season of 1949-50 three convection systems were investigated — a low-wall forced-air system, a high-wall forced-air system, and a gravity system, designated respectively as Series L-1, H-1, and G-1. Experimental conditions for this investigation are listed in Table 2.

1. *Forced-Air System with Low-Wall Registers (Series L-1)*

In this system (Fig. 7) the warm air was delivered to the bedrooms and to the bathroom through an insulated trunk duct in the attic. A high-wall register was used in the bathroom. Warm air for the living room was delivered directly from the furnace bonnet through a conventional wall stack to the low-wall register in the north wall of the living room. Return-air intakes were located above the doors of the bedrooms and high in the wall between the utility room and the dining area. The duct system was designed⁽¹²⁾ in accordance with *Manual 7*. In general, the operating conditions for the low-wall system were the same as for the perimeter system.

2. *Forced-Air System with High-Wall Registers (Series H-1)*

As shown in Fig. 8, warm air was delivered through a plenum above the furred ceiling in the hall to registers above the doors of the bathroom and bedrooms. The register for the living room was about 7 ft above the floor in the north wall of the living room. The return-air intakes in the hall and dining area were both located in the baseboard of the utility room walls. The registers and return-air intakes were sized⁽¹²⁾ in accordance with *Manual 7*, but the duct sizes were based on the requirements for the gravity system. Since the warm-air duct system was oversized for forced-air flow and since no changes were made in the blower speed, the resulting air-flow rate was higher than that for the low-wall system. The temperature rise through the furnace was 70 F under prolonged burner operation.

3. *Gravity System (Series G-1)*

For operation with the gravity system the blower and the division panel which separated the blower compartment from the upper part of

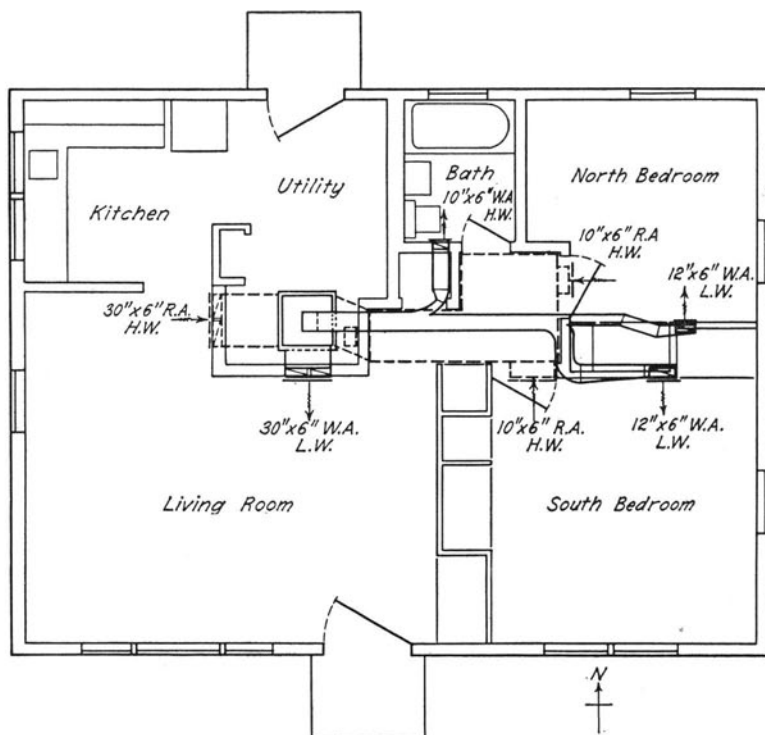


Fig. 7. Plan of Low-Wall Forced-Air System

the furnace were both removed from the furnace casing, and the conventional forced-air registers and return-air intakes were replaced with those suitable for gravity operation. The design of the duct system was based essentially on the procedure⁽¹³⁾ given in *Manual 5*.

In this discussion the results obtained with the three convection systems have been compared with those of Series P-5 for the two-loop perimeter system. Since most of the experimental data were obtained when the outdoor temperatures were between 30 and 40 F, the comparisons have been confined to this range, in which the blower operation was practically continuous for both the low-wall (Series L-1) and high-wall (Series H-1) systems.

15. Room-Air Temperatures

With the low-wall system, large cyclical variations in temperatures of the order of 3-4 F were obtained in the living zone, corresponding to variations in register-air temperature of 20 F. Similar temperature

variations were obtained at the ceiling level when the air was delivered in the upper part of the room through high-wall registers. However, with the high-wall system the temperature variations were smaller in the lower part of the room, amounting to about 1 F in the living zone. This compares favorably with the 0.5 F which was experienced with the perimeter system.

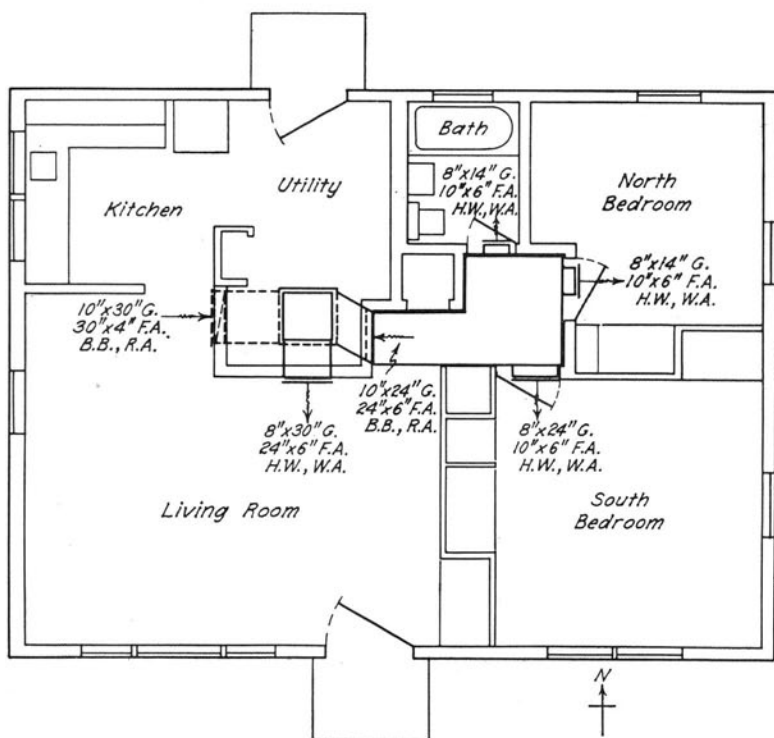


Fig. 8. Plan of Gravity System and High-Wall Forced-Air System

As shown in Table 3, the temperature balance between rooms at the sitting level was more satisfactory with the high-wall system than with either the low-wall or gravity systems. However, the degree of temperature balance obtained with the three convection systems was not as satisfactory as that for the perimeter system.

In the living room the temperature difference between the warmest and the coolest areas was about 2 F for both the gravity and high-wall systems, and was comparable with that obtained with the perimeter system. With the low-wall system, however, the delivery of warm air

Table 3
Temperature Differences Between Rooms for Two-Loop Perimeter
System and Three Convection Systems

All temperature differences shown were observed at the sitting level for an outdoor temperature of 30–40 F.

Series	Average Differences Between Living Room West and			Maximum Differences Observed Between	
	(1) S. Bedroom	(2) N. Bedroom	(3) Bath		
G-1 (Gravity)	2.8	3.3	3.7	Liv. Rm. W.—Bath	7.6
H-1 (High-Wall)	2.7	2.6	1.3	Liv. Rm. W.—S. Bedrm.	4.4
L-1 (Low-Wall)	1.3	2.5	2.2	Liv. Rm. E.—Bath	8.0
P-5 (Two-Loop Perimeter)	1.0	1.3	1.1	Liv. Rm. E.—N. Bedrm.	1.6

into the east half of the living room resulted in a larger difference in room-air temperatures between the east half and west half of the room.

The temperature differentials experienced with the various series are shown in Table 4, and typical room-air temperature gradients obtained at the west living room measuring station are shown in Fig. 9. Extremely large floor-to-ceiling temperature differentials were obtained with the high-wall (Series H-1) and gravity (Series G-1) systems, both of which utilized registers located near the ceiling. Furthermore, the floor-to-breathing-level differentials were also large, being of the order of 10 F. The panel heating effect of the warm ceiling gave a floor-surface temperature for the gravity system which was higher than that of the air 3 in. above the floor (see Fig. 9). In any event the room-air temperature differentials for the gravity and high-wall systems were not considered satisfactory. The temperature differentials obtained with the low-wall system showed a considerable improvement over those obtained with the high-wall and gravity systems, but were slightly larger than those for the perimeter system.

Considering both the temperature balance at the sitting level and also the room-air temperature differentials, it was concluded that the performances of the low-wall, high-wall, and gravity systems were less satisfactory than that of the two-loop perimeter system.

Table 4
Room-Air Temperature Differentials for
Two-Loop Perimeter System and Three Convection Systems
Outdoor temperature between 30 F and 40 F

Series	Between Floor Level and Breathing Level					Kitchen- Utility
	Liv. Rm. W.	Liv. Rm. E.	N. Bedroom	S. Bedroom	Bath	
G-1 (Gravity)	8.4	5.8	10.8	8.0	9.8	8.3
H-1 (High-Wall)	10.1	8.1	12.5	9.4	13.3	8.6
L-1 (Low-Wall)	6.9	6.3	6.1	5.5	8.6	7.3
P-5 (Perimeter)	4.9	1.7	4.3	4.9	4.0	4.1
Series	Between Floor Level and Ceiling Level					
	Liv. Rm. W.	Liv. Rm. E.	N. Bedroom	S. Bedroom	Bath	
G-1 (Gravity)	24.9	30.9	24.7	24.8	25.7	15.9
H-1 (High-Wall)	13.5	9.7	15.8	14.4	18.8	10.2
L-1 (Low-Wall)	7.7	6.7	6.9	5.7	20.3	10.9
P-5 (Perimeter)	5.7	3.2	5.9	8.0	10.9	11.1

16. Surface Temperatures

No appreciable differences were observed between the general patterns of floor-surface temperatures experienced with the low-wall, high-wall, and gravity systems, and the patterns were less satisfactory than that shown in Fig. 6 for the two-loop perimeter system. A comparison of floor-surface temperatures for the convection (Series L-1) and the perimeter (Series P-5) types of systems is shown in Fig. 10, in which temperatures at seven selected points have been plotted against indoor-

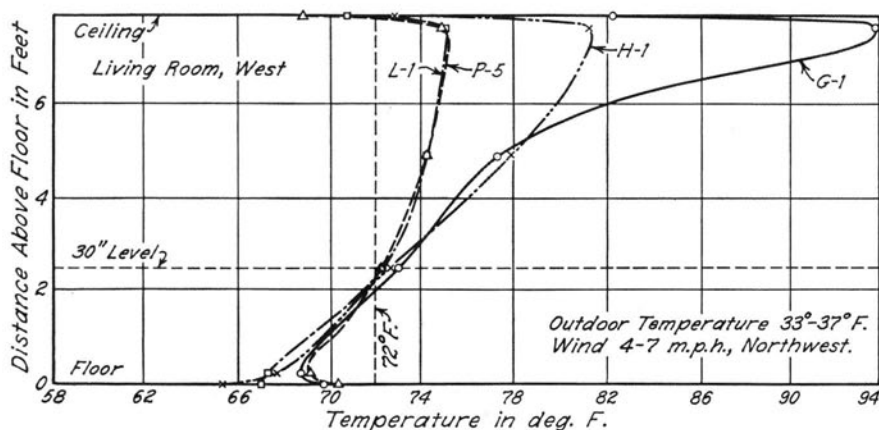


Fig. 9. Room-Air Temperature Gradients for Two-Loop Perimeter System and Three Convection Systems

outdoor temperature difference. In the central area of the house (station C-9) temperatures observed with the convection systems were approximately the same as those observed with the perimeter system. The difference in floor-surface temperatures was most apparent directly above the embedded ducts, but even at a distance of 3 ft from the outside walls (stations C-1, C-8, and C-12) the temperatures were also definitely lower with the convection systems. At the corners of the rooms 6 in. from each wall (stations F-15, F-19, and F-22), they were also definitely lower with the convection system than with the perimeter system. Although the floor-surface temperatures experienced with the perimeter system were not completely satisfactory, they were more satisfactory than those with any of the three convection systems.

The temperatures of floor, wall, and ceiling surfaces affect the radiation heat exchange between the occupant and the surfaces, and hence affect the comfort of the occupant. In this connection studies were made

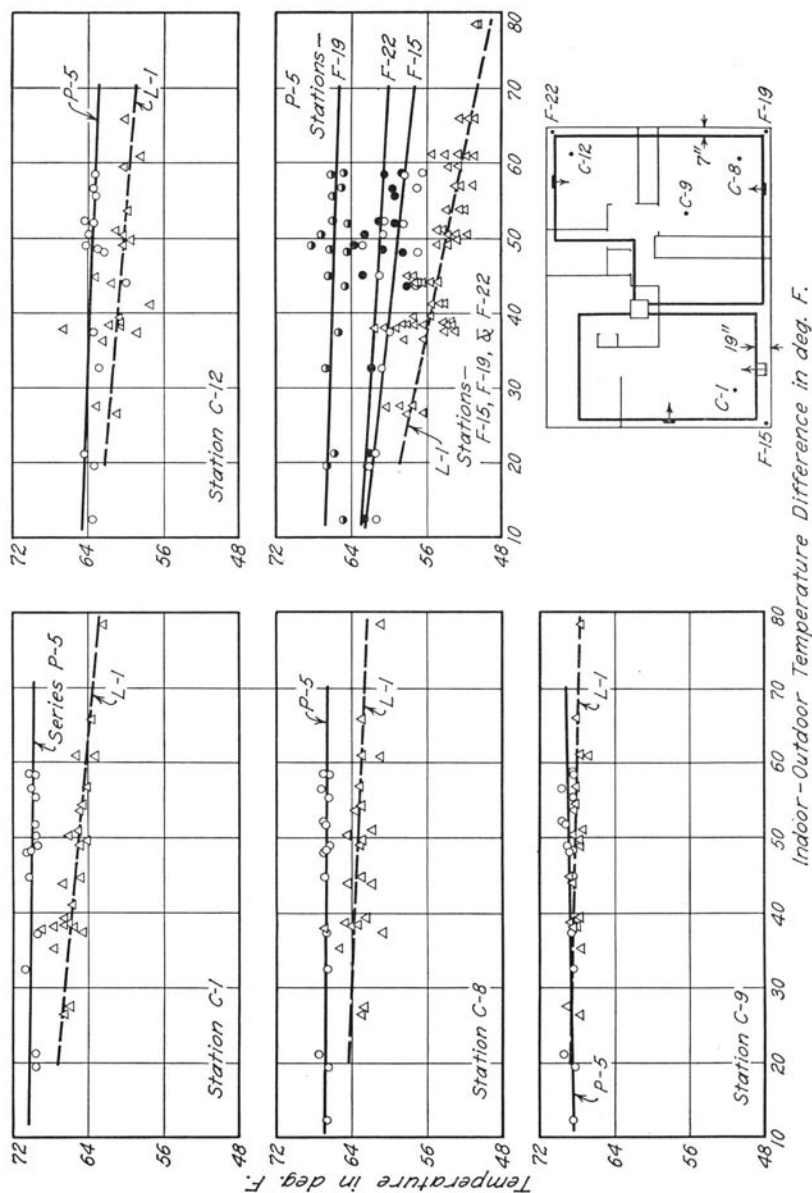


Fig. 10. Floor-Surface Temperatures as Affected by Indoor-Outdoor Temperature Difference

of the wall- and ceiling-surface temperatures in the Residence. At the sitting level the wall-surface temperatures were substantially the same for all four systems. In the upper part of the room, however, they showed trends similar to those of the ceiling-surface temperatures, which as indicated in Fig. 9 were substantially higher for the gravity system than for the high-wall system, as these in turn were higher than those for both the low-wall and perimeter systems. The average ceiling-surface temperatures, as observed at 20 stations in the Residence, were approximately 82 F for the gravity system, 76 F for the high-wall system, and 72 F for the low-wall and perimeter systems. The fact that the value for the gravity system was as high as 82 F indicates that a panel heating effect occurred which would tend to improve the comfort conditions. On the other hand, this value was obtained only by maintaining even higher air temperatures at the ceiling level.

In this connection it may be observed from curve G-1 in Fig. 9 that a ceiling-surface temperature of 82 F required an air temperature of about 94 F at the ceiling level. In other words, a large drop occurred across the air film adjacent to the ceiling surface. Hence it would appear that in order to obtain ceiling-surface temperatures which would provide an effective radiation panel, excessively high air temperatures at the ceiling level would be required. This in turn would result in excessive heat losses through the ceiling and particularly through the exposed wall and window surfaces. The resulting increase in fuel consumption would probably offset the gain in comfort resulting from the increased panel effects.

17. Average Daily Relative Humidities

The average daily relative humidities of the indoor air were a function of the outdoor-air temperature, but independent of the type of system under investigation. The trends shown in Table 5 were the same as those previously reported^(14, 7) for Research Residences No. 1 and No. 2.

Table 5
Humidity Data for Various Outdoor Temperatures

	Outdoor Temperature		
	50 F	30 F	10 F
Average Indoor Relative Humidity, percent	38	25	17
Average Indoor Humidity Ratio, lb of water per lb of dry air	0.0061	0.0042	0.0028
Average Outdoor Humidity Ratio, lb of water per lb of dry air	0.0060	0.0024	0.0009

The difference between the indoor humidity ratio and the outdoor humidity ratio was not large, primarily because the moisture input from domestic usage was negligible and the outdoor-air infiltration was relatively large.

18. Performance of Burner and Furnace

Complete performance data for the burner and furnace were obtained over a wide range of indoor-outdoor temperature differences. The data shown in Table 6 are averages obtained from the performance curves and are for a temperature difference of 35 F.

Table 6
Summary of Performance Data for Burner and Furnace (1949-50 Heating Season)
(Two-Loop Perimeter System and Three Convection Systems)

	<i>Experimental Conditions</i>			
	<i>G-1</i>	<i>H-1</i>	<i>L-1</i>	<i>P-5</i>
Air-Flow Rate, cfm	680	480	480
Temperature Rise Through Furnace, F	70	100	100
Fan-Switch Settings	Cut-in:100 Cut-out: 80	Cut-in: 100 Cut-out: 80	Cut-in: 100 Cut-out: 80
<i>Summary of Values for an Indoor-Outdoor Temperature Difference of 35 F</i>				
(a) Blower Operation, hr per day	21.4	23	23.0
(b) Blower Cycles, per 24 hr	24	10	5
(c) Electrical Input to Blower Motor, watt-hr per day	2400	2400	2450
(d) Bonnet-Air Temperature, F, av.	139	103	108	110
(e) Burner Operation, hr per day	10.8	9.0	8.0	9.0
(f) Burner Operations, per 24 hr	90	83	80	100
(g) Flue-Gas Temperature, F, av.	395	315	315	335
(h) Fuel Consumption, cu ft of gas per day	690	600	510	605
(j) Percentage Difference in Fuel Consumption from Series P-5	+14	-1	-16	0

With the exception of the fuel consumption (item h) and the accompanying burner operation (item e) the major items affecting performance of the forced-air systems were essentially similar. The most significant differences were observed in item h. These differences are best illustrated as percentages of the fuel consumption for the perimeter system (item j). The fuel consumption for the high-wall system was practically the same as that for the perimeter system. Apparently the additional ground-heat loss with the perimeter system was offset by the larger heat loss in the upper portions of the room with the high-wall system. The fuel consumption for the low-wall system was about 16 percent less than that for the perimeter system. Since the room-air temperatures for these two systems, as indicated in Fig. 9, were practically identical in the upper part of the room, the room-heat losses should be about the

same. Hence the difference in fuel consumption of 95 cu ft per day, shown by the values in item h, was attributed entirely to the additional heat loss through the floor slab with the use of a heated perimeter duct. This difference was equivalent to 35 Btu per hr per linear ft of exposed edge. Although a floor slab construction identical to that used with Series L-1 and P-5 was not investigated in the Floor Slab Laboratory, two constructions which approximated the floor slab in Residence No. 3 were studied.^(5, 11) The difference in heat loss through these two floor slabs was of the same order of magnitude as the value of 35 Btu per hr given above. It should be observed that in the case of the floor slab construction for Series L-1 the presence of the perimeter duct tended to provide an additional insulating effect at the edge of the slab. Consequently the 16 percent reduction in fuel consumption obtained with the low-wall system as compared with the perimeter system can be considered as the maximum that might be obtained.

The fuel consumption for the gravity system was approximately 14 percent greater than that for the perimeter system. The difference has been attributed to the increased heat loss through the upper part of the room as well as to the higher flue losses (item g) resulting from the reduced air-flow rates and high bonnet-air temperatures (item d).

V. PERFORMANCE OF RADIAL-FEEDER PERIMETER SYSTEM

19. Description and Procedure

This system consisted of an 8-in.-diam warm-air duct embedded in the concrete slab in the form of a single loop around the periphery of the floor with 8-in.-diam radial-feeder ducts connecting the subfloor plenum to the perimeter duct. A total of seven feeder ducts were installed and provisions were made for blocking each duct at both ends so that any combination of the seven could be studied. The feeders were 4 in. below the floor at the plenum and sloped upward to the junction with the perimeter duct, which was 2 in. below the floor. Dimensions of the subfloor plenum were 20 in. x 20 in. x 18 in. deep.

Four main series of studies—designated as P-11, P-12, P-13 and P-14—were conducted; the duct arrangements are shown in Fig. 11. The three-feeder arrangements of Series P-11 and P-12 were considered as minimum installations for a structure of the size of Residence No. 3, whereas the arrangement used in Series P-14 (five feeders) was considered as an optimum. The objective of this study was to determine the general performance characteristics of a radial-feeder perimeter system, and specifically to determine the optimum number and location of feeder ducts. Experimental conditions are listed in Table 2.

In addition to the changes in the duct arrangement, the following changes were made in the conversion from the two-loop system to the radial-feeder system:

- a. A down-flow furnace was used in place of the up-flow furnace.
- b. The bathtub was raised 2 in. above the floor, and the warm air for the bathroom was introduced from the perimeter duct into the space below the tub. The air then entered the room through a narrow grille at the base of the tub.
- c. Floor registers 4 in. x 14 in. were substituted for the larger floor registers. In addition, the number of registers was increased.

20. Uniformity of Room-Air Temperatures at Sitting Level

The cyclical variation of room-air temperatures was negligible in all four series, amounting to less than 0.5 F. Also, the response of the system to sudden changes in outdoor temperatures was considered to be satis-

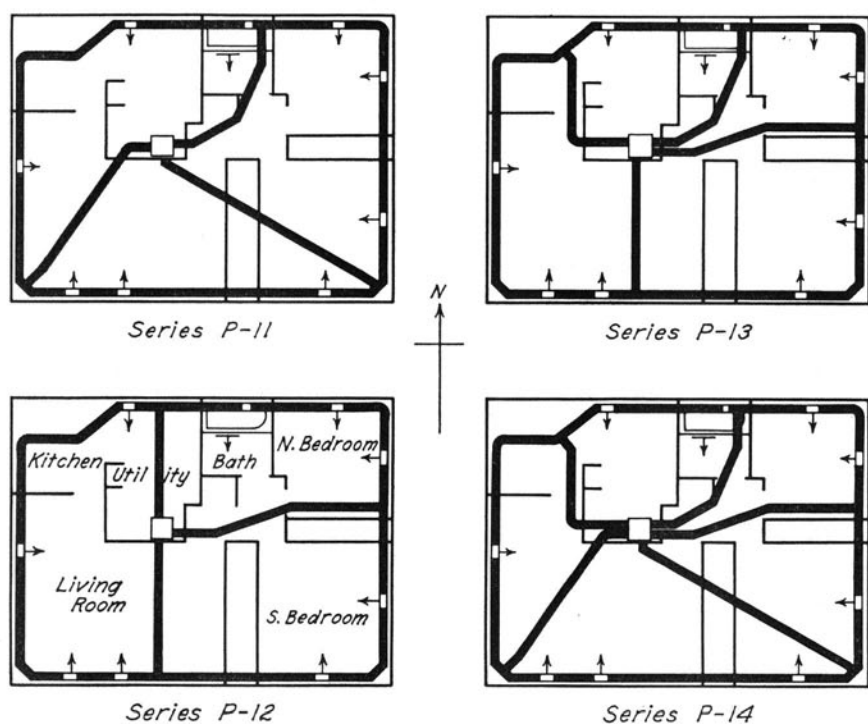


Fig. 11. Four Arrangements of Radial-Feeder Perimeter-Loop System

factory, since no evidence of lag or overrun of room-air temperatures was obtained during periods of rapidly changing outdoor temperatures.

In contrast to the experience with the two-loop perimeter system, little difficulty was encountered in balancing the radial-feeder system. By merely adjusting the shutter dampers in the floor registers, the temperature in any room in the Residence could be readily raised or lowered. In Series P-12 and P-13, for which corner feeder ducts were not used, the balancing process required more care than in either Series P-11 or P-14. (See Section 22.)

The best balance was obtained with Series P-14, in which five feeder ducts were used. The temperature difference between the living room and the two bedrooms was consistently less than 1 F, whereas with the other three series the temperatures of the bedrooms were generally 1—2 F lower than those in the living room. Differences of 1—2 F are comparable to those obtained with the two-loop system but could have been reduced further by manipulation of the dampers. This was not the case with the two-loop system, for which a practical limit of adjustment had been reached.

The temperature difference between the warmest and the coolest areas of the living room was only 1 to 2 F for all four series of studies. In those arrangements in which corner feeder ducts were not used (Series P-12 and P-13) the air temperatures near the corner of the room were 1 to 1.5 F lower than for those arrangements utilizing corner feeder ducts (Series P-11 and P-14). This difference indicates the desirability of running feeder ducts into the corner areas.

21. Room-Air Temperature Differentials

The room-air temperature differentials experienced with the radial-feeder perimeter system were satisfactory with all four of the duct arrangements studied, and were superior to those obtained in the studies previously reported in this bulletin. As shown in Fig. 12, the temperature gradients at seven measuring stations in the Residence for an outdoor temperature of about 20 F were slightly better for the five-feeder arrangement (Series P-14) than for the other three series. This was attributed to the larger panel heating effect resulting from the use of a larger number and more effective location of feeder ducts. Figure 13 compares the temperature gradients for the two-loop system (Series P-5) and the radial-feeder system (Series P-13) for an outdoor temperature of about 10 F. The arrangement for Series P-13 was selected for the comparison because it was most nearly comparable in number and location of feeder ducts to the two-loop arrangement.

Comparison of the differentials (Figs. 5, 12, and 13) for the two-loop and radial-feeder perimeter systems permits the following generalizations.

a. Delivery of warm air into rooms from low-wall or baseboard registers resulted in differentials which were markedly superior to those obtained with high-wall delivery. (See Fig. 13. The bathroom differential from floor to ceiling levels was 16.6 F for Series P-5; it was reduced to -3.4 F for Series P-13.)

b. Locating the feeder ducts so that all areas of the living space were provided with a relatively direct supply of warm air resulted in lower differentials. (See Fig. 13, north bedroom for both series. Also see south bedroom.)

c. The use of two floor registers in a room, rather than one, resulted in better diffusion and lower register-air velocities, which in turn resulted in lower differentials. (See Fig. 13. Compare the cases shown for the south bedroom.)

d. Locating the feeder ducts so that they extended into exposed corners of the floor provided an additional heat source in areas which tended to be cool and resulted in more satisfactory differentials in those areas.

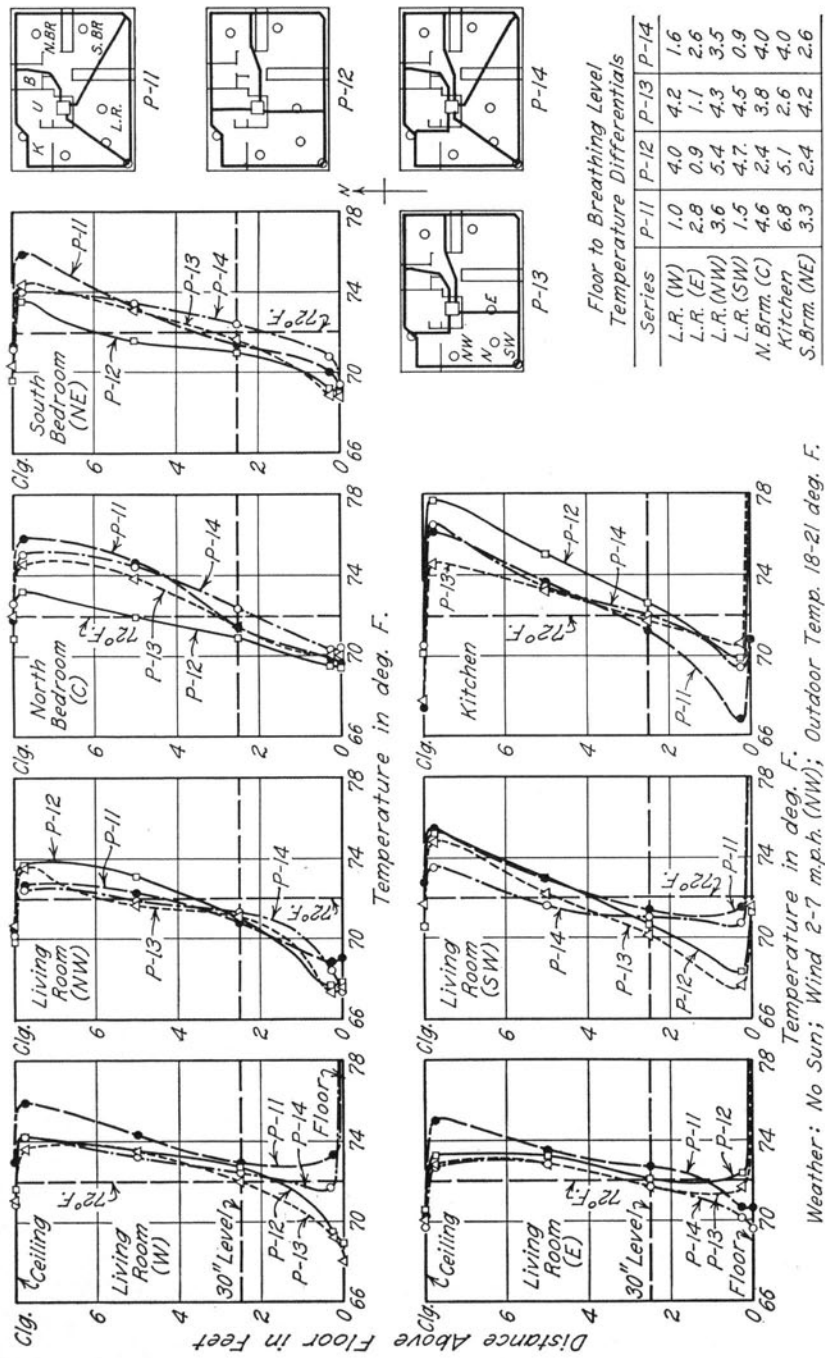
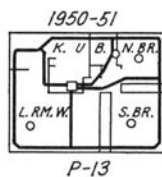
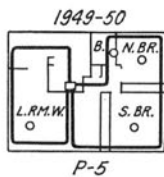
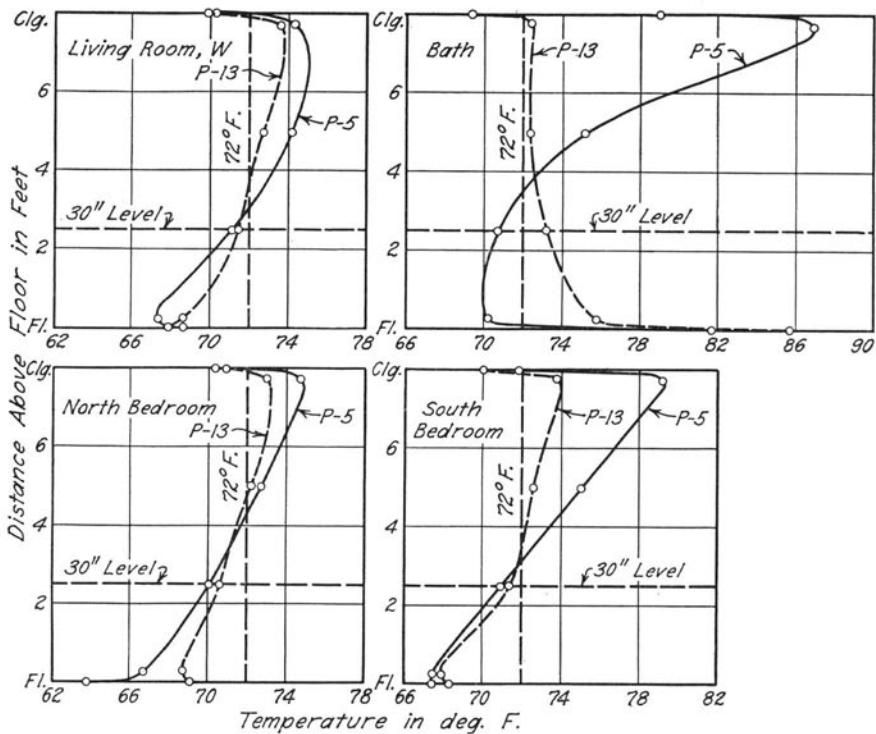


Fig. 12. Room-Air Temperature Gradients for Radial-Feeder Perimeter-Loop System



Room-Air Temperature Differentials

	3" Floor to 30" Floor		3" Floor to 60" Floor		3" Floor to 3" Ceiling	
	P-5	P-13	P-5	P-13	P-5	P-13
L.R.(W)	3.8	2.8	6.8	4.1	7.0	5.0
Bath	0.6	2.6	5.0	3.4	16.6	3.4
N. BR.	3.4	1.9	6.0	3.5	8.1	4.2
S. BR.	3.5	3.5	7.6	4.7	11.8	5.9

Fig. 13. Room-Air Temperature Gradients for Radial-Feeder and Two-Loop Perimeter Systems

(See Fig. 12. Compare the difference between Series P-11 and P-12 for east living room with the difference between the same series for west living room.)

e. Locating the registers below windows resulted in lower differentials, because the warm air mixed with the downward current of cool air from the windows. (See Fig. 12, Series P-11. Compare kitchen and northwest living room.)

f. The use of a larger number of feeder ducts generally resulted in improved differentials, since the panel heating effect was increased. (See Fig. 12. Compare Series P-11 and P-14.) However, the location of the

feeder ducts was more important than the number used, provided that the number was sufficient to assure an adequate delivery of warm air to all sections of the perimeter duct. (See Fig. 12. Compare Series P-11 and P-13.)

The effect of indoor-outdoor temperature differences on the room-air temperature differentials for the three-feeder arrangement used in Series P-11 is shown in Fig. 14 for three typical locations. In the southwest living room and the north bedroom the temperature differentials increased only slightly with an increase in indoor-outdoor temperature difference. In the kitchen, however, the differential increased to a larger extent, since a register was not located beneath the window to offset the effects of cool air from cold window surfaces and infiltration. On the whole, the change in differential with a change in outdoor temperature was small — in fact, smaller than those observed with a conventional warm-air system in an insulated residence⁽⁷⁾ provided with storm sash.

22. Floor-Surface Temperatures

The floor-surface temperatures obtained with Series P-14 and P-11 of the radial-feeder system were considerably better than those obtained with the two-loop system, but those for arrangements in which the feeder ducts did not extend into the corners (Series P-12 and P-13) were not markedly better.

As indicated in Section 20, attempts to attain desirable floor-surface temperatures with Series P-12 and P-13 involved difficulties in obtaining a good balance of room-air temperatures. With these series the floor-surface temperatures were low in both the southwest corner of the living room and the southeast corner of the south bedroom. By suitable adjustments of the register dampers it was possible to increase the air-flow rate through the corner sections of the perimeter duct and thereby increase the floor-surface temperatures in those corners. However, the increase in the flow rate through the corner sections was accomplished only by reducing the flow rate through the register nearest the feeder duct, which in turn made it difficult to supply sufficient heated air to the room to maintain the desired room-air temperature. Furthermore, the reduction in flow rate through the nearest register resulted in excessive air velocities through the register farther down the duct. Typical floor-surface isotherms are shown in Fig. 15 for Series P-11. These isotherms represent conditions at the end of a 3-day period during which the outdoor temperature was about 30 F, cloudy weather prevailed, and the wind was approximately 10 mph from the west. Although a more uniform pattern

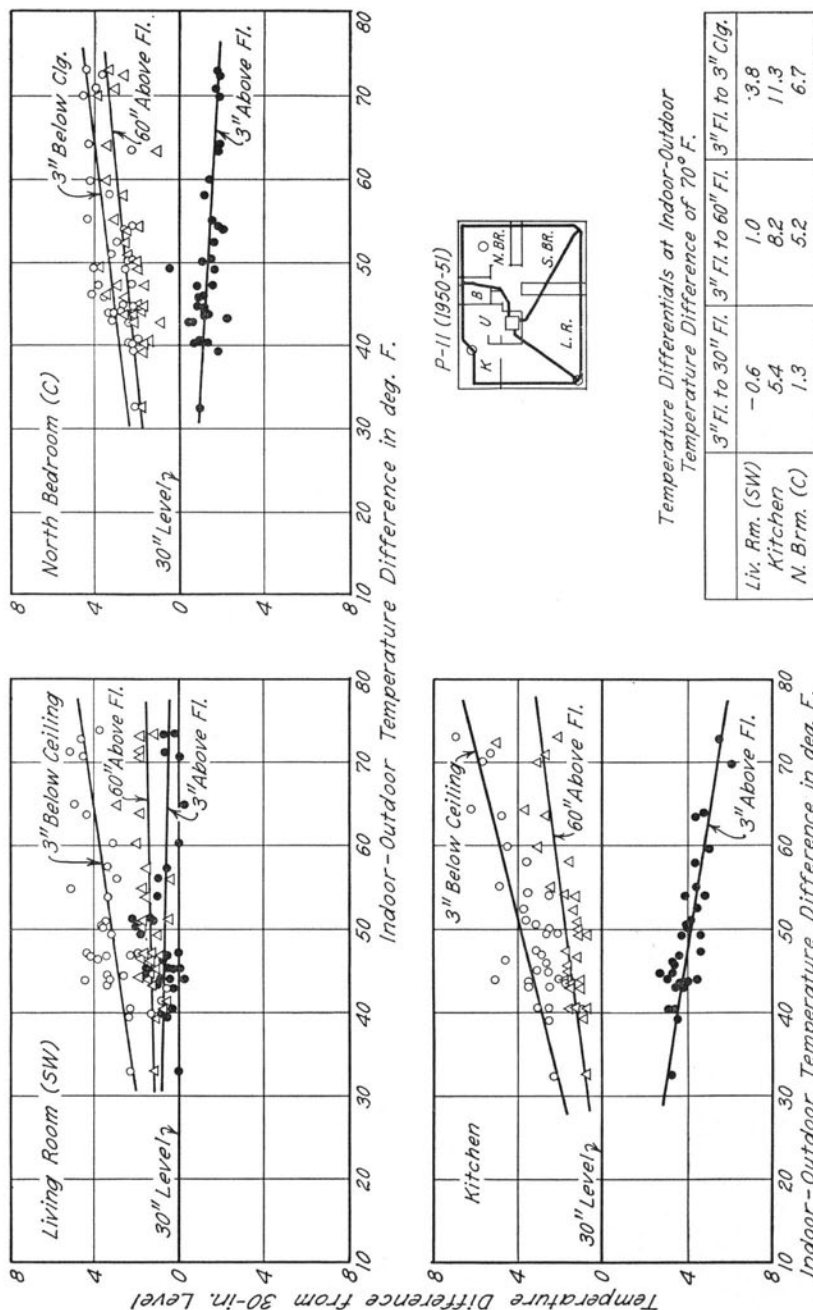


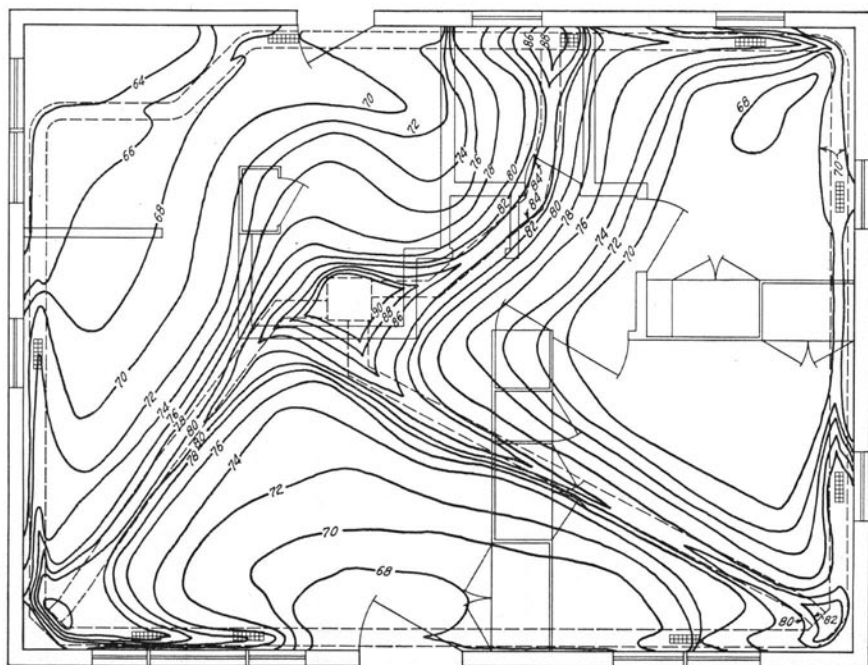
Fig. 14. Room-Air Temperature Differentials as Affected by Indoor-Outdoor Temperature Difference

of temperatures was obtained with the five-feeder arrangement (Series P-14) than with Series P-11, the pattern for the simpler three-feeder arrangement is shown, since it is more representative of the type of perimeter systems applicable to small houses.

In connection with Fig. 15 the following observations were made.

a. The floor-surface temperatures in the kitchen were low, because the distance along the feeder and perimeter ducts from the subfloor plenum was great. However, since most of the floor area was devoted to cabinets and was not normal living space, the temperatures were not considered to be critically low. This low-temperature area does indicate the need for extending feeder ducts into corner areas as well as for having sections of the perimeter duct at relatively short air-flow distances from the subfloor plenum. With Series P-14 a feeder duct was extended into the corner of the kitchen; the resulting floor-surface temperatures were considerably improved over those shown in Fig. 15.

b. The practically parallel isothermal lines over the feeder ducts were typical for a perimeter system in which the feeder ducts are installed with a uniform upward pitch from the subfloor plenum to the perimeter



Weather: No Sun; Outdoor Temp. 30°-35° F; Wind 9-12 mph (SW)

Fig. 15. Floor-Surface Isotherms with Radial-Feeder Perimeter-Loop System (Series P-11)

duct. The depth varied from about 4 in. at the furnace to 2 in. at the junction with the perimeter duct. This temperature pattern closely resembles that obtained from the studies in the Floor Slab Laboratory,⁽¹⁰⁾ in which the pitch was 1 in. in 5 ft.

c. For outdoor temperatures of 30 to 35 F the maximum floor-surface temperatures did not exceed 85 F, except for an area within about 2 ft of the subfloor plenum. Under average winter weather conditions for the locality, therefore, the floor-surface temperatures in the entire living area were well below the accepted⁽⁴⁾ maximum value of 85 F.

From an analysis of the floor-surface temperatures experienced with the two-loop system and each of the radial-feeder arrangements over a wide range of indoor-outdoor temperature differences, the following generalizations were made.

a. As the outdoor temperature decreased, the floor-surface temperatures in those areas away from the embedded ducts, and in areas near the perimeter duct but at a considerable distance from the subfloor plenum, did not change appreciably from those shown in Fig. 15. However, surface temperatures exceeding 85 F were obtained in three localized areas:

- (1) The temperature of the floor near the furnace increased considerably with the higher duct-air temperatures which accompanied low outdoor temperatures. For example, the surface temperatures directly above the feeder ducts and 15 in. from the subfloor plenum increased to 110 F when the bonnet-air temperature was near the maximum of 170 F. The surface temperatures decreased sharply, however, with an increase in distance from the subfloor plenum, so that the temperature above the feeder ducts attained a maximum of only 95 F at a distance of 5 ft from the plenum. Furthermore, at a distance of 15 in. from the plenum and 18 in. from the nearest feeder duct the floor-surface temperature reached only 85 F.

- (2) Beyond a distance of 5 ft from the plenum, surface temperatures directly above the feeder ducts attained values of 95–100 F at an outdoor temperature of –10 F. Also, temperatures in excess of 85 F existed for a distance of less than 15 in. on each side of the centerline of the ducts.

- (3) Warm floor surfaces were obtained directly above the tee connections that joined the feeder duct to the perimeter duct. For an outdoor temperature of –10 F the surface temperature was about 100 F.

b. From the standpoint of comfort as affected by floor-surface temperatures the commonly accepted⁽⁴⁾ maximum values are 85 F for living spaces and 120 F for borders and aisles. Since the surface temperatures

observed near the furnace and over the tee fittings were less than 120 F for borders, neither of these areas was considered critical. It is true that a narrow strip of floor above the feeder ducts attained temperatures in excess of the 85 F accepted as a maximum for living areas. However, since such temperatures were obtained only in severe weather and since the critical area was narrow, exceeding these accepted limits was not considered serious.

23. Performance of Burner and Furnace

As has been said (page 33), complete performance data for the burner and furnace were obtained over a wide range of indoor-outdoor temperature differences. The general performance characteristics of the burner and blower were essentially the same for all four duct arrangements, since the operating conditions of the furnace were identical. However, a substantial difference in fuel consumption was observed as between the radial-feeder system and the two-loop system. The larger fuel consumption with the two-loop system was attributed to the fact that the furnace was designed to operate as either a gravity or a forced-air unit, whereas the furnace for the radial-feeder system was a down-flow unit and was designed specifically for forced-air operation. The furnace for the two-loop system operated at higher flue gas temperatures and a lower CO₂ content in the flue gas. In addition, this up-flow furnace required a connecting duct between the furnace bonnet and the subfloor plenum. The vagrant heat gain from this duct and from the bonnet caused the kitchen and utility room to overheat and reduced the amount of heat available to other rooms in the house. This in turn required increased operation of the burner in order to satisfy the heat demands of the other rooms.

Actual fuel consumption was compared with the theoretical requirements. For instance, the fuel consumption curve for the radial-feeder system indicated an average consumption of 1175 cu ft per day (49,000 Btu per hr) at an indoor-outdoor temperature difference of 80 F and an average wind velocity of only 7.5 mph. For these same weather conditions the calculated heat loss was 43,200 Btu per hr and the corresponding theoretical input rate was 54,000 Btu per hr for an assumed bonnet efficiency of 80 percent. This difference of approximately 5000 Btu per hr between the actual and the theoretical input rates was attributed to the vagrant heat gains from the furnace casing and vent pipe and the direct heat gains from lights, etc.

On the assumption that this difference of 5000 Btu per hr for vagrant and direct heat gains would also occur under design conditions (51,600 Btu per hr heat loss for an indoor-outdoor temperature difference of 80 F and 15 mph wind velocity), a margin of safety amounting to 8 per-

cent of the theoretical input rate of 65,000 Btu per hr would be provided. Hence the practice of selecting the fuel input rate on the basis of the total calculated heat loss divided by the assumed bonnet efficiency appears to be reasonable.

In connection with the blower performance, measurements were made of the pressure losses for the duct system for Series P-11 and were compared with calculated values (Table 7). The values shown in column 1 represent calculated pressure losses and summarize the data given in Appendix A. Though a discrepancy was noted between the estimated pressure loss through the registers (0.007 in. of water) and the observed pressure loss (0.025 in.), the total pressure losses for the warm-air duct

Table 7
Pressure Losses in Radial-Feeder Perimeter System (Series P-11)

	Calculated Values From Appendix A (1)	Observed Values* (2)
1. Total Air-Flow Rates, cfm		
a—Return Air	478	500
b—Supply Air	555†	500
2. Pressure Losses Through Warm-Air Duct System, in. of water		
a—Feeder Duct	0.057	‡
b—Perimeter Duct	0.004	‡
c—Registers	0.007¶	‡
d—Total	0.068	0.025
3. Pressure Losses Through Return-Air Duct System, in. of water		
a—Return-Air Duct	0.005	‡
b—Return-Air Intake	0.018	‡
c—Total	0.023	0.028
4. Total Pressure Available External to Blower, in. of water	0.091	0.098

* Measurements made with no heat supply to the furnace.

† Calculations based upon average duct-air temperature with system operating with a temperature rise through the furnace of 100 F.

‡ Values not measurable.

¶ Estimated value based on wide-open register.

system were practically the same. A reasonably good agreement was also shown in the corresponding values for the return-air duct system (item 3c) and for the total system (item 4). Considering the numerous assumptions made in the calculation procedure and the fact that the air-flow rates (items 1a and 1b) were not in agreement, the correlation between the actual and calculated values was good. This was particularly true when the calculated pressure losses for the ducts (items 2a, 2b, and 3a) were adjusted to the actual flow rates and the calculated register loss (item 2c) was arbitrarily increased to the measured value of 0.025 in. The calculation procedure given in Appendix A should therefore be applicable to other duct-air velocities and to other duct sizes. In any event, since the total loss of the entire duct system was only of the order of 0.10 in. and since the blower-furnace unit was rated to discharge air against an external static pressure of 0.20 in., it was apparent that the warm-air ducts could have been reduced in size.

VI. SUMMARY AND CONCLUSIONS

Research Residence No. 3 was built in order to permit comparison of the performances of several warm-air heating systems which appeared to be suitable for heating basementless houses having concrete slab floors. The investigation included a study of the following five types of warm-air heating systems:

1. a two-loop perimeter system
2. a radial-feeder perimeter system
3. a conventional forced-air system with low-wall registers
4. a conventional forced-air system with high-wall registers, and
5. a gravity system.

The Residence was a low-cost basementless house with a concrete slab floor laid on the ground. The inside dimensions were 24 ft x 32 ft and the corresponding floor area was 768 sq ft. The design heat loss was 51,600 Btu per hr, including a ground loss of 9150 Btu per hr. The total design heat loss was equivalent to 67 Btu per hr per sq ft of floor area.⁽¹⁵⁾ The walls were not insulated and storm sash were not used. The Residence was considered, therefore, to be a typical low-cost home. The 4-in.-thick floor slab was provided with a moisture barrier and with $2\frac{5}{32}$ -in. edge insulation. The heat transmission factor for the floor under design conditions was assumed to be 80 Btu per hr per linear ft of exposed edge.

The essential features of the five heating systems and an evaluation of their relative merits are presented in the following discussion.

1. *Gravity System.* The furnace was located in the utility room and was connected to a horizontal plenum which served four high-wall registers. The two return-air intakes were located in the baseboard. The performance of this system was not satisfactory, because of high air temperatures at the ceiling level and low temperatures of the floor surface. The fuel consumption was also relatively high.

2. *High-Wall Forced-Air System.* Four high-wall registers and two return-air intakes in the baseboard were used. The performance showed some improvement over that for the gravity system as far as room-air temperature differentials were concerned, but no improvement was observed in the floor-surface temperatures.

3. *Low-Wall Forced-Air System.* Three low-wall registers, one high-wall register in the bathroom, and three high-wall return-air intakes were

used. An improvement was noted in the room-air temperature differentials from floor to ceiling. None, however, was observed in the temperatures of the floor surface. Fuel consumption was reasonably low.

4. *Two-Loop Perimeter System.* In this system 8-in.-diam sheet metal ducts were embedded 2 in. below the top of the concrete slab and were installed in the form of two loops which enclosed a major portion of the floor area. The warm air was delivered into a subfloor plenum, from which it was forced outward to the perimeter duct and then to floor registers below the windows.

The fuel consumption for the two-loop perimeter system was about 18 percent greater than for the low-wall system. This difference was attributed to the additional heat loss through the floor slab with the use of a heated perimeter duct. A considerable improvement was observed in the room-air temperature differentials and the floor-surface temperatures. However, a concentrated source of panel heating effect existed in that portion of the living room floor above the parallel feeder ducts, contributing to an unbalance of the system. The conclusion was reached that the feeder ducts should be spaced far apart and if possible under separate rooms. Furthermore, the evidence indicated that a more favorable design would consist of a continuous perimeter duct with (a) relatively short distances between the feeder ducts and the registers, (b) the feeder ducts so arranged that all sections of the perimeter duct would serve as effective air passages, and (c) relatively high air-flow rates maintained in all sections of the perimeter duct.

5. *Radial-Feeder Perimeter System.* This system consisted of an 8-in.-diam warm-air duct embedded in the concrete slab in the form of a single loop around the periphery of the floor, and a series of 8-in.-diam radial-feeder ducts from the subfloor plenum to the perimeter duct.

The response of the system to sudden changes in outdoor temperature was satisfactory: no evidence of lag or overrun of room-air temperatures was obtained during periods of rapidly changing outdoor temperatures. The conventional room thermostat used with the system proved to be satisfactory for controlling the room-air temperature.

Room-air temperature differentials with this system were low; even at an outdoor temperature as low as 0 F the temperature differentials from the floor level to the ceiling level were less than about 6 F throughout most of the house. This is less than the temperature differentials normally experienced with a conventional high-wall system in a well-insulated house with a full basement.

The practice of balancing the system by adjusting the shutter dampers on the floor registers proved satisfactory. In contrast to the experience

with the two-loop perimeter system no difficulty was encountered in balancing the radial-feeder system. As shown in Appendix A, the actual pressure losses for the warm-air and return-air duct systems were in reasonably good agreement with the calculated values. Furthermore, the pressure loss external to the furnace-blower combination was less than the 0.20 in. of water available. This finding indicated that some reduction in the size of feeder and perimeter ducts could be made without exceeding the available pressure of the blower.

Floor-surface temperatures were satisfactory for the duct arrangements in which feeder ducts extended into the exposed corners of the floor. In cases where feeders were not extended into the corner areas the floor-surface temperatures tended to be low. Floor-surface temperatures in excess of the commonly accepted maximum value of 85 F for living areas were experienced. However, these temperatures occurred only in a narrow section of floor directly above the feeder ducts; hence they were not considered critical. In no case did temperatures above the perimeter duct or in areas along walls adjacent to the subfloor plenum exceed 120 F commonly accepted as the maximum for borders and aisles.

The operating conditions for the system followed the continuous air circulation principle. That is, the differential setting of the room thermostat was adjusted to provide short cycling of the burner during mild weather, the fuel-input rate was adjusted to a value equal to the heat loss of the house divided by the assumed bonnet efficiency, the blower was adjusted to provide a temperature rise through the furnace of 100 F, and the fan switch settings were such as to cause the blower to operate practically continuously. These operating conditions proved to be satisfactory for the perimeter system.

In connection with the fuel-input rate a comparison of the actual fuel consumption with the theoretical requirements indicated that there was a margin of safety of 8 percent in determining the theoretical input rate. This difference of 8 percent between the actual and the theoretical requirements indicates that the practice of selecting the fuel-input rate on the basis of the total calculated heat loss divided by the assumed bonnet efficiency is reasonable. A heat balance for the radial-feeder perimeter system is given in Appendix B.

On the basis of the results obtained with the four separate arrangements for the radial-feeder system it was evident that such a system should be so designed that feeder ducts extend into exposed corners of the floor, that the total air-flow distances from subfloor plenum to registers are relatively short, and that relatively high air-flow rates are maintained in all sections of the perimeter duct.

APPENDIX A

CALCULATED PRESSURE LOSSES FOR RADIAL-FEEDER PERIMETER SYSTEM

The following summary gives the essential details of the simplified procedure followed in calculating pressure losses for the radial-feeder perimeter system. The duct arrangement for Series P-11 has been used.

1. The design loss, including subfloor loss, was calculated as 51,600 Btu per hr.

2. The design bonnet-air temperature was assumed to be 170 F and the return-air temperature 70 F. Therefore the design temperature rise through the furnace was 100 F.

3. The air-flow rate in the return-air duct was 478 cfm at a temperature of 70 F. This value is shown in Table 7, item 1a.

4. The average length of the feeder ducts was 18 ft. The average temperature drop through the feeder ducts was assumed to be 30 F, giving an average feeder duct-air temperature of 155 F. The total air-flow rate in all the feeder ducts at a temperature of 155 F was 555 cfm. This value was also used for the total air-flow rate entering the perimeter duct. (See Table 7, item 1b.)

5. The average air-flow rate through each of the three feeder ducts was assumed to be one-third of 555 cfm, or 185 cfm. The average flow through any section of perimeter duct was assumed to be one-half of 185 cfm, or 93 cfm.

6. The pressure loss for the 8-in.-diam feeder duct was determined from the air friction chart⁽⁴⁾ as 0.065 in. of water per 100 ft. The total equivalent length of feeder duct (including subfloor plenum entrance, a 90-deg elbow, and tee junction) was 88 ft. The pressure loss in the feeder duct was 0.88×0.065 , or 0.057 in. of water. (See Table 7, item 2a.)

7. The average length of perimeter duct from the tee junction to the midpoint of the perimeter duct included between two junctions was 20 ft. The pressure loss in the perimeter duct was determined from the air friction chart⁽⁴⁾ as 0.018 in. of water per 100 ft. The pressure loss for this 20 ft section was 0.20×0.018 , or 0.0036 in. of water. (See Table 7, item 2b.)

8. Since the perimeter duct size was constant the duct was considered to function as an extended plenum. Hence it was assumed that the static pressures would be practically constant at all sections.

9. The average air-flow rate through each of the nine registers was assumed to be 63 cfm. Since no data were available on pressure losses through a floor register at right angles to the direction of flow in the main duct, the available information on a register in a stackhead was used. For this purpose the data from Reference 16 (p. 42, Table II) for a 14-in.-x-4-in. register were used. The register resistance was 0.051 in. for a stack velocity of 500 fpm. For the average stack velocity of 180 fpm the register loss was considered to be 0.007 in. for a wide-open register. If dampering were used, the value of 0.007 in. would be low. (See Table 7, item 2c.)

10. The equivalent length of the 8-in.-x-20-in. return-air duct, including one 90-deg shortway elbow, was 17 ft. The pressure loss for the duct for an air-flow rate of 478 cfm was 0.03 in. of water per 100 ft, or 0.005 in. for the 17-ft length. (See Table 7, item 3a.)

11. From Reference 16 (p. 22, Fig. 6) the pressure loss for a bar register having 75 percent free-area opening was shown as 2.0 velocity pressures. The pressure loss for the 30-in.-x-8-in. return-air intake at a free area velocity of 382 fpm was considered to be 0.018 in. (See Table 7, item 3b.)

A comparison of these calculated pressure losses and the measured pressure losses is given in Section 23 and Table 7.

APPENDIX B

HEAT BALANCE FOR RADIAL-FEEDER PERIMETER SYSTEM*

A detailed study was made of the heat balance for the radial-feeder perimeter system (Series P-11 and P-12). The heat gains and heat losses for the Residence were computed from established heat transfer equations and special measurements. Since the calculations were too extensive to reproduce here, only a summary of the final results for Series P-11 is presented. The data in Table 8 show calculated and measured values for the components of heat balance for a day in which the outdoor temperature was 35 F, a 10-mph wind velocity prevailed, and no solar heat gain occurred. The calculated values for the subfloor heat loss and the above-ground heat loss were determined in accordance with the usual methods for calculating the heat losses of a structure. The total calculated heat loss was 21,700 Btu per hr. Assuming a flue-gas loss of 20 percent, which would be equivalent to 5450 Btu per hr, the theoretical required input rate (to be supplied by the furnace) was found as 27,150 Btu per hr.

The following observations were made from an analysis of the calculated and measured values presented in Table 8.

1. The calculated value for subfloor loss (item 1) as based upon a heat-loss factor of 45 Btu per hr per linear ft of exposed edge of the floor slab was low. This was attributed to the fact that the factor of 45 Btu per hr was determined in the Floor Slab Laboratory⁽¹¹⁾ for a 1-in.-thick insulation extending 18 in. downward along the edge of the slab, whereas in Residence No. 3 the actual insulation used was only $2\frac{5}{32}$ in. thick and extended downward to a depth of only 12 in.

2. The calculated value for the above-ground heat losses was in excellent agreement with the measured values.

3. The net flue-gas loss as measured at the point where the flue pipe left the living space was only 3970 Btu per hr. When the heat regains from the flue pipe (item 2d) and the furnace casing (item 2c) were taken into account, the total of 5440 Btu per hr was practically identical with the calculated value of 5450 Btu per hr. The calculated value was based upon the conditions at the flue outlet of the furnace.

4. Although the total measured input rate to the Residence was 26,350 Btu per hr, the heat supplied by the furnace amounted to only

* The material in this Appendix is condensed from a thesis, "Heat Balance of a Radial-Feeder Perimeter Heating System," submitted by G. W. Sadler in 1951. A similar study was conducted by J. E. Sjoldal in 1950 with the two-loop perimeter system.

Table 8

Components of Heat Balance for Radial-Feeder Perimeter System (Series P-11)

Values based on outdoor temperature of 35 F and wind velocity of 10 mph. Observed values determined on Feb. 16, 1951, during which wind was from the southwest and no solar heat gain occurred.

<i>Items</i>	<i>Calculated Values, Btu per hr</i>	<i>Measured Values, Btu per hr</i>
1. Subfloor Loss	5 050	6 280
2. Above-Ground Loss	16 650	16 100
Offset by:		<i>Subtotals</i>
a. Warm air from floor registers		9 350
b. Panel heating effect		3 090
c. Furnace casing heat transfer		1 100
d. Flue pipe heat transfer		1 370
e. Electrical energy		1 190
f. Energy from observers		1 000
3. Subfloor and Above-Ground Losses	21 700	22 380
4. Flue Gas Loss	5 450	3 970
5. Total Input Rate	27 150	26 350
6. Supplied by Furnace	27 150	23 750

23,750 Btu per hr. The difference was attributed to the electrical energy input (item 2e) and the heat gain from observers (item 2f).

5. The only unavailable energy was the net flue-gas loss as determined at the ceiling level where the flue pipe left the living space. This amounted to 16.7 percent of the fuel-input rate to the furnace. In other words, the over-all efficiency of the furnace in this installation can be considered to be 100.0 - 16.7, or 83.3 percent.

6. The value for the panel heating effect from the heated floor slab (item 2b) was about one-third as large as that for the energy entering the rooms through the registers (item 2a). In other words, the radial-feeder system functioned primarily as a convection system but to some extent as a panel heating system. The total of items 2a and 2b was 12,440 Btu per hr, which represented about 77 percent of the heat supplied to offset the above-ground heat loss.

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